Available online at www.sciencedirect.com
ScienceDirect
Energy Procedia 85 (2016) 550 – 558
Energy Procedia

Sustainable Solutions for Energy and Environment, EENVIRO - YRC 2015, 18-20 November 2015, Bucharest, Romania

Numerical prediction of thermal comfort and condensation risk in a ventilated office, equipped with a cooling ceiling

Catalin Teodosiu[a]*, Viorel Ilie[a], Raluca Teodosiu[a]

[a]*Technical University of Civil Engineering, 122 – 124 Lacul Tei Bvd, Bucharest 020396, Romania*

Abstract

In the current context of more and more energy efficient buildings, performant HVAC systems are required. The solution based on radiant cooling ceiling systems has proven successful in terms of high energy efficiency and good levels of thermal comfort. Nevertheless, there is an important issue concerning the cooling ceilings: the risk of condensation. As a result, the aim of this study is to investigate the thermal comfort and condensation risk for buildings with ventilation systems and cooling ceilings. The approach is based on CFD (Computational Fluid Dynamics) technique, introducing special methodologies in order to deal with convection and diffusion phenomena for water vapor, required for comprehensive thermal comfort analyzes and condensation mechanisms. The numerical model is applied in the case of a small office, taking into account a radiant cooling ceiling over the entire ceiling surface and two configurations of ventilation systems: mixing ventilation and displacement ventilation. Several simulations have been completed for different conditions concerning the air flow rates of the ventilation systems and temperatures of the cooling ceiling. It has been concluded that mixing ventilation has superior behavior, both in terms of thermal comfort and condensation risk, when warm and moist air is supplied in the room. On the other hand, the results show that when untreated air is supplied by the ventilation system (no matter its configuration, mixing or displacement), thermal comfort in the office cannot be properly assured only by radiant cooling ceiling systems.

Peer-review under responsibility of the organizing committee EENVIRO 2015
Keywords: CFD modeling; thermal comfort; cooling ceiling; condensation risk

* Corresponding author.
E-mail address: cteodosiu@yahoo.com

1876-6102
Peer-review under responsibility of the organizing committee EENVIRO 2015
doi:10.1016/j.egypro.2015.12.243

1. Introduction

World primary energy consumption was about 12.5 billion tons of oil equivalent in 2013, which means an increase of 2.3% over the previous year and even if global energy consumption increased only by 0.9% in 2014 [1], it is estimated that by 2040, the world energy consumption will grow by over 37% compared to 2010 [2].

In this context, the buildings sector represents the main consumer since buildings consume 40% of total primary energy across European Union (EU) and nearly half (47.6%) of all energy produced in the United States [3]. At the same time, the buildings sector is an important factor in greenhouse gas (GHG) emissions as it is responsible for 36% of total GHG emissions [4]. In addition, the buildings sector is constantly increasing. As a result, it is expected that by 2050 the total number of buildings in the EU will grow by over 25% [3], which will result in increased energy consumption in this sector.

Regarding the energy consumption of buildings, the largest amount of energy (more than 60%) in EU is used for thermal comfort (heating and cooling) [3]. Consequently, there is a great potential to significantly reduce the energy consumption of buildings by using energy-efficient HVAC (heating, ventilation, and air conditioning) systems.

In line with this, the emerging technique based on radiant cooling ceiling systems have proven more energy efficient than conventional air conditioning systems. Accordingly, simulations carried out by Stetiu [5] have estimated HVAC savings in cold, moist climates to be 17% to 42%, and an average savings of 30% in warm, dry climates.

Recent studies have also revealed the benefits of the radiant cooling ceiling systems related to thermal comfort issues of long-time staying occupants: less perceptible air movement and more uniform distribution of air temperature [6-8]; reduced vertical temperature difference and decreased draft risk [9,10].

Taking into account all these benefits of energy saving and improved thermal comfort, radiant cooling ceiling systems have been extensively studied lately. Investigations have been conducted through both experimental [7-10] and numerical [7,9,10] studies.

On the other hand, all analyses dealing with cooling ceilings are focused either on energy saving issues or on thermal comfort aspects. However, there is an important feature related to the functioning of radiant cooling ceilings systems: the risk of condensation. This issue is not thoroughly investigated as it is assumed that the basic correlation surface temperature – dew point of the air (based on simple psychrometric chart calculations for moist air) is sufficient. But how do we proceed when we do need to use lower ceiling temperatures in order to increase the cooling load and, in the same time, humid air is supplied for ventilation purposes? How should we proceed in these conditions in order to avoid the condensation without affecting thermal comfort? Could we optimize the correlation “dew point temperature – ventilation air flow rates – air movement in the room” to obtain comfortable environment by lowest possible surface temperature without generating condensation? This study attempts to bring elements of response to these problems by using CFD (Computational Fluid Dynamics) technique for a case study.

2. Airflow and condensation modeling

Taking into consideration the objectives of this study (comprehensive analyze of thermal comfort and investigation of risk condensation), an integrated moisture – energy – air flow approach is required. This is accomplished through a CFD model with the following main characteristic: an equation expressing the conservation of the water vapor mass fraction is added to the basic equations governing a turbulent confined non-isothermal airflow. This allows modeling of transport and diffusion phenomena for water vapor. The conservation equation for the water vapor can be written in the following general form:

$$\rho \frac{\partial}{\partial x_i}\left(u_i m_{i'}\right) + \frac{\partial}{\partial x_i} J_{i',i} = S_{i'} \tag{1}$$

where the left-hand side terms stand for the convective term (ρ - density, x_i - spatial coordinate, u_i – velocity component in i direction, $m_{i'}$ - water vapour mass fraction) and diffusion term respectively ($J_{i,i'}$ – water vapour diffusion flux), while the right-hand side term $S_{i'}$ represents source terms.

The diffusion term in Eq. (1) considers both molecular and turbulent diffusion phenomena. The molecular diffusion is classically solved by Fick's first law (water vapor diffusion flux is proportional to the concentration gradient), while the turbulent diffusion is taking into account by means of the turbulence model used to describe the flow of the humid air in the CFD simulation.

The main features of the CFD model developed according to the above methodology are presented in Table 1. More details about this CFD model can be found in [11].

Table 1. CFD model

Aspect	Description
Fluid	Air – water vapor mixture
Flow	Three-dimensional, steady, non-isothermal, turbulent
Computational domain discretization	Finite volumes, unstructured mesh (tetrahedral elements)
Turbulence model	Shear Stress Transport (SST) turbulent kinetic energy-specific turbulent dissipation rate (k-ω), with low-Reynolds corrections
Radiation model	Discrete ordinates (DO)
Numerical resolution	Second-order upwind scheme
	Velocity-pressure coupling: SIMPLE algorithm
	Convergence acceleration: algebraic multigrid

Regarding the condensation model, this is developed using the methodology proposed by International Energy Agency [12]. As a result, the condensed vapour density flux is computed based on the assumption that the water vapor transport in air is mainly due to convective mechanisms, the exception being nearby the surfaces where the diffusion has the main role:

$$\Phi_{vap.cond} = \beta(P_{vap.air} - P_{vap.surface}) \tag{2}$$

where β (s/m) - proportionality coefficient that describes the water vapor diffusion between the indoor air and the walls surface, $p_{vap.air}$ - vapor pressure in air, and $p_{vap.surface}$ - vapor pressure on the wall surface.

In addition, the coefficient β from Eq. (2) can be directly correlated with the convective heat transfer coefficient for applications in the field of thermal building [12]. As a result, the coefficient β in Eq. (2) continually varies with the convective heat transfer coefficient, resulting from heat exchanges calculation between the fluid and the ceiling in the CFD simulations.

The convective heat transfer coefficient is computed in the model taking into account the triangular faces (on the walls) of the discretization mesh and the centers of the first tetrahedral discretization elements within the indoor air (viscous zone), in the case of an unstructured mesh. Consequently, for the viscous sublayer (e.g. cell with a triangle of area S_F and temperature T_F), the convective heat exchange takes the form:

$$\Phi = h_c(T_w - T_{air})S = h_c(T_F - T_{C0})S_F \tag{3}$$

where T_w – wall surface temperature, T_{air} – air temperature, and S – surface; T_{C0} – first barycenter tetrahedron (near the wall) temperature.

On the other hand, the viscous region near the ceiling is characterised in terms of heat exchanges by conduction, the convection playing a negligible role. Consequently, the heat flux density can be taken into account by the classical Fourier law, which means that the convective heat transfer coefficient can be determined as follows:

$$\varphi = -\lambda_{air}\,\overrightarrow{grad T}\,\vec{n} = \lambda_{air}\left(\frac{\partial T}{\partial n}\right) = h_c(T_F - T_{C0}) => h_c = \frac{\lambda_{air}}{(T_F - T_{C0})}\left(\frac{\partial T}{\partial n}\right) \tag{4}$$

where λ_{air} - thermal conductivity of the humid air and the temperature gradient is calculated in the following manner:

$$\left(\frac{\partial T}{\partial n}\right)=\frac{(T_F-T_{C0})}{S_F}\alpha \tag{5}$$

$$\alpha=\frac{\overrightarrow{S_F}*\overrightarrow{S_F}}{\overrightarrow{S_F}*\overrightarrow{s_0}}$$

where S_F represents the area of a triangle and α:
with s_0 = (coord_F – coord_{C0})

Based on Eq. (4), the value of the convective heat transfer coefficient is determined, for a discretization cell that has one face on the ceiling, by taking into account the following data: temperature difference between the center of the triangle (on the ceiling) and the centroid of the tetrahedron which contains the triangle; distance between the center of the triangle and the centroid of the tetrahedron enclosing the triangle; the surface of the triangle; thermal conductivity of the moist air located in the tetrahedron centroid [11].

Once the coefficient β is solved (based on its correlation with the convective heat transfer coefficient –see above), the mass flow rate of water vapor condensed on the ceiling $m_{vap.cond}$ is calculated, for each discretization triangle that is positioned on the ceiling, as follows [11]:

$$\dot{m}_{vap.cond}=\frac{dm_{liq.surface}}{dt} \tag{6}$$

$$
\begin{aligned}
&\textit{if}\\
&\quad P_{vap}-P_{vap.sat}>0\\
&\qquad \dot{m}_{liq.surface}=7{,}4\text{x}10^{-9}h_c S_{F_i}(P_{vap.}-P_{vap.sat})\\
&\textit{else}\\
&\qquad \dot{m}_{liq.surface}=0
\end{aligned}
$$

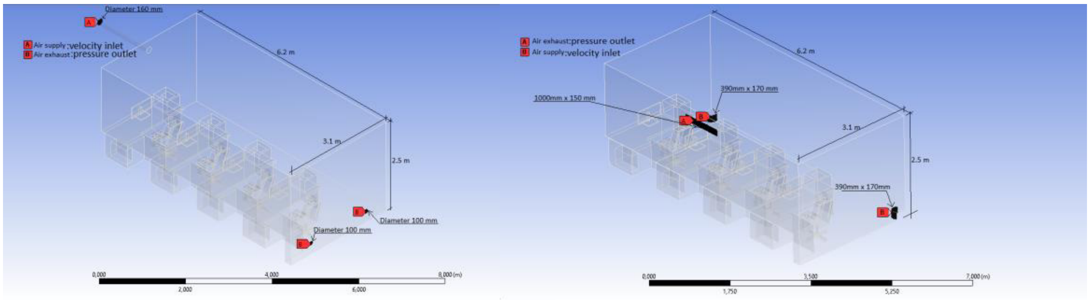
where mliq.surface - condensed vapor flux, based on Eq. (2), pvap.sat - saturation pressure of water vapor, and pvap. - partial pressure of water vapor.

Finally, using the mass flow rate of water vapor condensed on the surface, the link with the volume computation should be completed by taking into account the mass balance and the energy balance [11]. The mass balance is carried out in the following way: the condensate flow rate is removed by means of source term that is introduced in the water vapor conservation equation of the CFD model, Eq. (1). On the other hand, the energy balance is based on sink terms added in the energy conservation equation of the CFD model. These terms are related to latent heat of vaporisation and sensible heat of the removed condensed water vapour [11].

The airflow model and condensation model have been developed using the general-purpose, finite-volume, Navier-Stokes solver Ansys Fluent (version 15.0.0). In addition, the condensation modeling has been integrated in the CFD simulations using specific code facility (UDF - user defined functions).

3. Case study

The numerical model presented above is applied in this work in the case of a small office (6.2 x 3.1 x 2.5 m^3) for four persons, taking into account a radiant cooling ceiling over the entire ceiling surface and two configurations of ventilation systems: mixing ventilation system (air supply at the upper part of the room and air exhaust at the lower part of the room, see Fig. 1a) and displacement ventilation system (air supply in the occupied zone and air exhaust at the upper part of the room, see Fig. 1b).



a) b)

Fig. 1. Case study : a) Mixing ventilation; b) Displacement ventilation

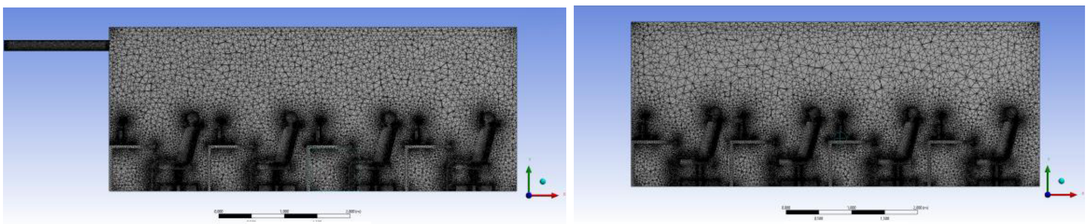
The geometrical representation of the human body is based in this study on simplified shapes: head – sphere (20 cm. diameter), neck - cylinder (5 cm. height and 5.5 cm radius), body and hands – parallelepiped (35 x 17 x 60 cm^3, legs – two cylinders (90 cm. height and 5.6 cm radius each). The thermal equilibrium of the human body has been taking into account by constant temperature fixed on the whole body surface (33ºC).

Six simulations have been performed for the conditions given in Table 2. Boundary conditions are shown in Fig. 1.

Table 2. Case study

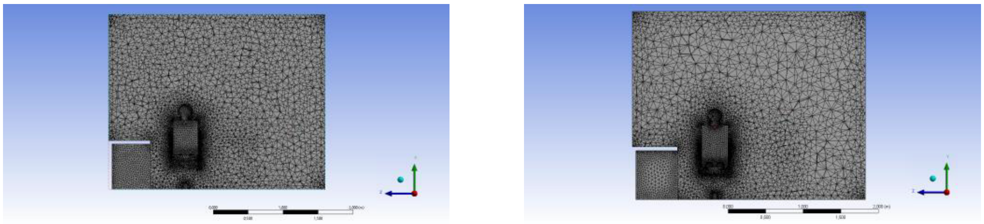
Test	Air changes per hour *(h^{-1})*	Inlet air temperature *(ºC)*	Inlet moisture content *(g/kg)*	Ceiling temperature *(ºC)*
M1 - Mixing ventilation	1	35.0	10.5	17
M2 - Mixing ventilation	3	35.0	10.5	17
M3 – Mixing ventilation	3	35.0	10.5	14
D1 – Displacement ventilation	1	35.0	10.5	17
D2 – Displacement ventilation	3	35.0	10.5	17
D3 – Displacement ventilation	3	35.0	10.5	14

The mesh for the whole computational domain discretization contains 5,788,819 cells for the configuration with mixing ventilation (Fig. 2a) and 6,007,026 cells for the configuration with displacement ventilation (Fig. 2b), respectively. These discretizations have been used as a result of numerical mesh sensitivity tests carried out to obtain grid independent solutions. In addition, the discretization was refined in the regions with strong flow gradients. This can be seen in Fig. 3. Based on this improvement technique of the grid in the zone of boundary layers, we have achieved the following minimum values of the non-dimensional wall distance (y^+): 0.11256 for mixing ventilation and 0.08156 for displacement ventilation. These values ensure the mesh suitability for near solid boundaries treatment of the low-Reynolds flow.



a) b)

Fig. 2. Mesh section in the median vertical plane of the enclosure : a) Mixing ventilation; b) Displacement ventilation



a) b)

Fig. 3. Mesh detail around occupants : a) Mixing ventilation; b) Displacement ventilation

4. Results

Among the indexes used to predict the thermal comfort conditions, the Predicted Mean Vote (PMV) has been successfully employed in buildings equipped with HVAC systems [13]. Consequently, the thermal comfort is approached within our study based on PMV index, as this has been defined in [14].

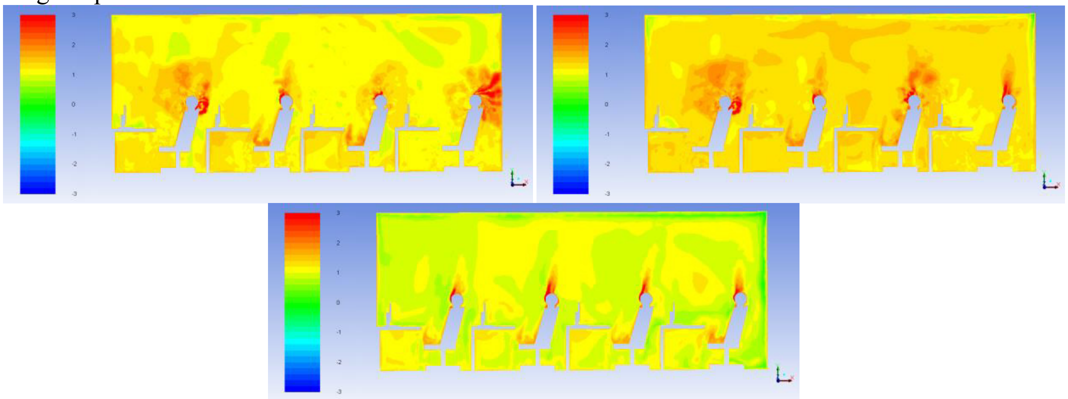
It is worthwhile to mention that the numerical assessment of environmental indices has included all the parameters required in the PMV expression. As a result, air velocity and air temperature are determined from the CFD flow equations, while the mean radiant temperature is calculated based on the temperatures of walls (including the cooling ceiling temperature – see Table 2). On the other hand, in order to take into consideration the indoor air humidity influence on thermal comfort, the water vapor pressure is computed all over the computational domain by means of the humidity model introduced in the global CFD approach, using the water vapor mass fraction given by Eq. (1). In addition, the simulations have taken into account the moisture released from the occupants. The value of the total humidity generation rate (respiration and transpiration) for one person was 60 g/h. This value corresponds to the metabolic rate of an average adult who performs typical office work. Related to this, the values of personal parameters are set in the PMV calculations as it follows: activity level, 1.2 met for the rate of metabolic heat production and 0 met for the rate of mechanical work, while the value of clothing thermal resistance is supposed to be 0.9 clo.

The PMV results are reported in the center plane of the occupants, 45 cm. from the median vertical plane of the office, normal to the air terminal devices.

Concerning the configuration with mixing ventilation combined with cooling ceiling, Fig. 4 shows the results for configurations M1, M2, and M3, respectively (see Table 2). It can be seen that the most comfortable environment in the occupied zone of the office takes place in the case M3. This highlights the contribution of the cooling ceiling to ensure adequate thermal comfort conditions despite supplying by the ventilation system of an important warm air flow rate. On the other hand, this can be provided by lowering the ceiling temperature (from 17ºC to 14ºC) but this can lead to a ceiling temperature below the dew point of indoor air which means condensation. The results obtained concerning the benefits of the coupling of primary air system with cold radiant ceiling panels in terms of thermal comfort are confirmed by other studies [9].

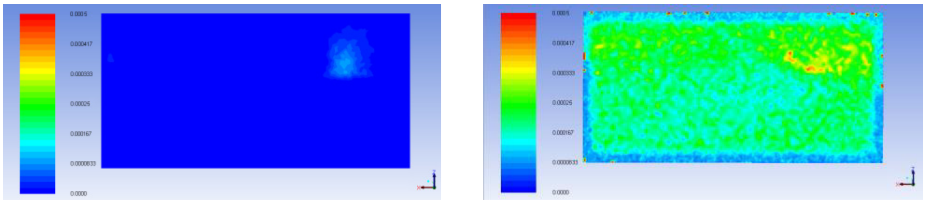
The risk of condensation is shown in Fig. 5. Compared to the situation when the ceiling temperature was higher (17ºC), and where there is only an extremely limited area with very low quantity of condensate (Fig. 3a), the configuration with lower ceiling temperature causes significant condensation almost on the entire ceiling surface (Fig. 3b) in spite of good levels of PMV in the occupied zone of the room. These results illustrate the situations in which thermal comfort cannot be provided by radiant cooling ceiling systems because there is a pronounced risk of condensation due to the introduction of fresh air (warm and moist) for ventilation.

The same remarks can be noticed in the case of configuration with displacement ventilation, with the mention that PMV values indicate a much warmer indoor environment (Fig. 6) and the risk of condensation is much higher (Fig. 7) compared to the mixing ventilation. This is explained by different air flow that occurs in this case in the office due to different position of air supply and air exhaust (in this case the hot air is supplied directly in the occupied zone while in the other case there is a well-known mechanism of mixing, leading to fast air temperature homogeneity). In fact, the results achieved show that displacement ventilation combined with cooling ceiling is not recommended in this case because it does not ensure thermal comfort and the risk of condensation is high even for high ceiling temperatures.



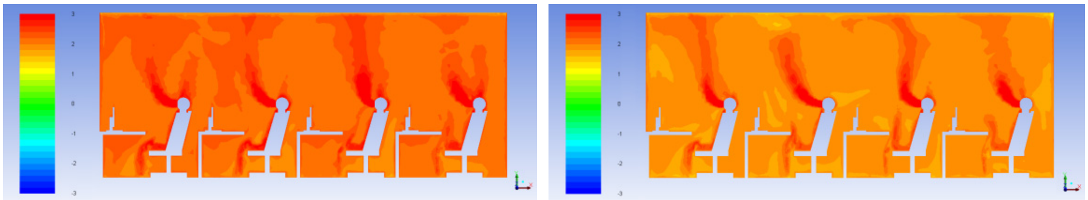
M3 (3 h^{-1}; ceiling temperature: 14ºC)

Fig. 4. Mixing ventilation system with cooling ceiling: PMV values

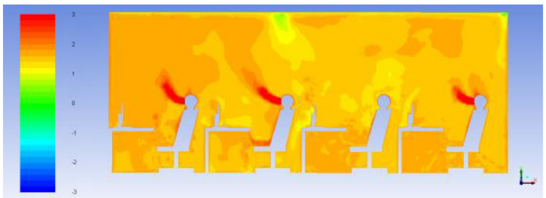


a) M2 (3 h^{-1}; ceiling temperature: 17ºC) b) M3 (3 h^{-1}; ceiling temperature: 14ºC)

Fig. 5. Mixing ventilation system with cooling ceiling: distribution and amount of condensate on the ceiling (l)

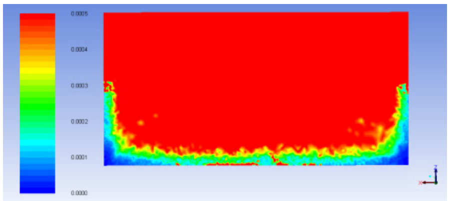


D1 (1 h^{-1}; ceiling temperature: 17ºC) D2 (3 h^{-1}; ceiling temperature: 17ºC)

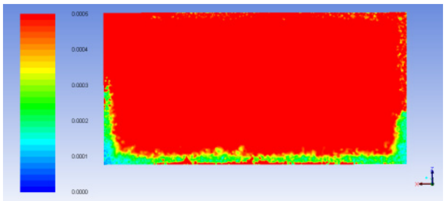


D3 (3 h^{-1}; ceiling temperature: 14ºC)

Fig. 6. Displacement ventilation system with cooling ceiling: PMV values



a) D2 (3 h^{-1}; ceiling temperature: 17ºC)



b) D3 (3 h^{-1}; ceiling temperature: 14ºC)

Fig. 7. Displacement ventilation system with cooling ceiling: distribution and amount of condensate on the ceiling (l)

On the other hand, the results show that, although cooling ceiling systems may generally allow higher indoor air temperature for the same thermal comfort sensation [15], in the case of supplying untreated air (warm and humid air) by the ventilation system (no matter its configuration -mixing or displacement), acceptable conditions of thermal comfort cannot be achieved only by radiant cooling ceiling solutions.

5. Conclusions

This work has numerically investigated the thermal comfort and risk condensation in a small ventilated office, equipped with a cooling ceiling. The CFD model developed in this study is therefore able to evaluate the thermos-convective field in ventilated rooms provided with cooling ceiling. In addition, the numerical model allows the prediction of moisture transport in enclosures. As a result, this approach can be used to assess the main environmental parameters: air velocity, air temperature, and air humidity. For this reason, the model presented here can be particularly useful in investigations on thermal comfort for enclosures provided with HVAC systems. On the other hand, the numerical models allows also predicting the amount of condensate on cold surfaces (e.g. cooling ceilings), the exact position and its evolution in space and in time. This feature of the model can have multiple applications on how to optimize the cooling load of radiant systems in order to avoid the condensation, but without affecting thermal comfort, for different ventilation schemes.

In this context, the case study taken into account in this study has revealed that better results can be achieved using mixing ventilation than displacement ventilation, both in terms of thermal comfort and condensation risk. This emphasizes the strong influence of the air movement on heat and mass (humidity) transfer phenomena that take place in ventilated rooms equipped with cooling ceilings.

Finally, the results of this work show that when warm and moist air (untreated air) is supplied by the ventilation system, thermal comfort cannot be appropriately provided only by radiant cooling ceiling systems.

Acknowledgements

This work was supported by a grant of the Romanian Ministry of Education and Research, CNCS-UEFISCDI, project number PN-II-ID-JRP-RO-FR-2012-0071.

References

[1] BP. BP Statistical Review of World Energy 2015. Retrieved September 10, 2015 from: http://www.bp.com/en/global/corporate/about-bp/energy-economics/statistical-review-of-world-energy.html

[2] Conti J, Holtberg P, Napolitano S, Schaal AM. International Energy Outlook 2014. Washington DC: U.S. Energy Information Administration (EIA); 2014. Retrieved October 5, 2015 from: http://www.eia.gov/forecasts/ieo/pdf/0484(2014).pdf

[3] Carutasiu MB. Contributions to the development of gray–box mathematical model used for analyzing high energy efficiency buildings (in Romanian), PhD. Thesis, University Politehnica of Bucharest; 2015

[4] Mlecnik E. Innovation Development for highly energy-efficient housing: Opportunities and challenges related to the adoption of passive houses. Amsterdam, The Netherlands: IOS Press (Delft University Press); 2013.

[5] Stetiu C. Radiant cooling in U.S. office buildings: towards eliminating the perception of climate-imposed barriers. PhD. Thesis, University of California, Berkeley; 1997
[6] Novoselac A, Srebric J. A critical review on the performance and design of combined cooled ceiling and displacement ventilation systems. Energy Build. 2002;34(5):497–509.
[7] Kim T, Kato S, Murakami J, Rho JW. Study on indoor thermal environment of office space controlled by cooling panel system using field measurement and the numerical simulation. Build. Environ. 2005;40(3):301-310.
[8] Memon RA, Chirarattananon S, Vangtook P. Thermal comfort assessment and application of radiant cooling: a case study. Build. Environ. 2008;43(7):1185–1196.
[9] Corgnati SP, Perino M, Fracastoro GV, Nielsen PV. Experimental and numerical analysis of air and radiant cooling systems in offices. Build Environ. 2009;44(4):801–806.
[10] Catalina T, Virgone J, Kuznik F. Evaluation of thermal comfort using combined CFD and experimentation study in a test room equipped with a cooling ceiling. Build Environ. 2009;44:1740–1750.
[11] Teodosiu R. Integrated moisture (including condensation) – energy – airflow model within enclosures. Experimental validation. Build. Environ. 2013;61:197-209.
[12] International Energy Agency (IEA). Energy conservation in buildings and community systems programme. IEA – Report Annex 14, vol 1;1991.
[13] Orosa JA, Oliveira AC. A new thermal comfort approach comparing adaptive and PMV models. Renewable Energy 2011; 36:951-956.
[14] International Standard Organization. ISO 7730 - Moderate thermal environments-determination of the PMV and PPD indices and specifications of the conditions for the thermal comfort. Geneva, 1994.
[15] Kim J, Tzempelikos A, Braun JE. Review of modelling approaches for passive ceiling cooling systems. Journal of Building Performance Simulation 2015;8(3):145-172.